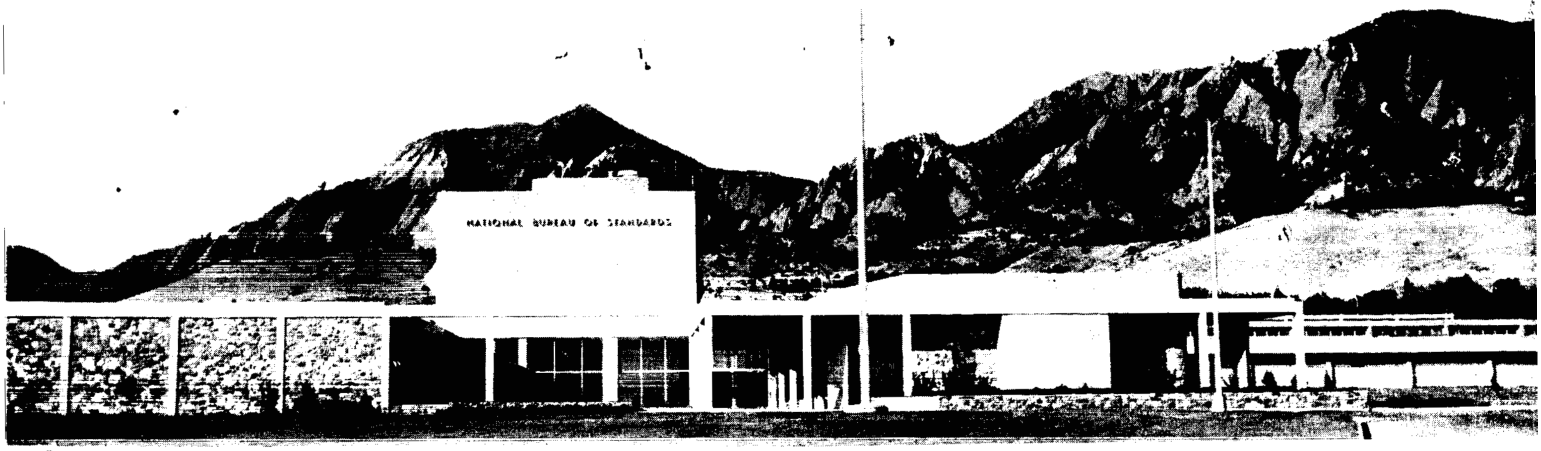

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NBS REPORT

8444

SATELLITE REFRIGERATION STUDY
PART I--TECHNICAL ANALYSIS

P. C. Vander Arend, D. B. Chelton, and D. B. Mann

NBS

U. S. DEPARTMENT OF COMMERCE
NATIONAL BUREAU OF STANDARDS
BOULDER LABORATORIES
Boulder, Colorado

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Technical Report

to

National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, Maryland

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U.S. DEPARTMENT OF COMMERCE

NATIONAL BUREAU OF STANDARDS

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SATELLITE REFRIGERATION STUDY

1.0 Introduction

This report is a review and analysis of the state-of-the-art of miniature refrigeration. The work was performed at the request of NASA Goddard Space Flight Center as an effort to determine present availability of a low temperature refrigeration system for satellite mounted infrared sensor cooling. To provide background for the analysis, a brief review of possible thermodynamic systems is made.

The analysis deals with both open and closed cycles and components. It outlines fundamental problems, some of which are a function of the size of the component. On the basis of this analysis, predictions may be made as to the possible application of the various systems. A number of organizations were visited in order to evaluate the progress made by these companies, and to discuss technical details and objectives.

A separate section deals with the subliming refrigerator, which basically consists of a storage dewar filled with a solid cryogen. A review of the state-of-the-art indicates that up to the present, little work has been done by industry. Fundamentally, the subliming refrigerator appears very attractive for use in a satellite because of the zero power requirements and the high reliability which may be expected. The state-of-the-art in cryogenic engineering is sufficiently advanced to permit the construction of a reliable subliming refrigerator with the expenditure of moderate amounts of money and time. This is not true for some of the mechanical refrigerators now under development.

2.0 Mechanical Refrigerators

A considerable effort is underway in development of miniature refrigeration equipment for applications similar to the one required for the Nimbus satellite. Most of this work has been sponsored by the Department of Defense, and more specific information about the various program is given under the description of efforts by individual companies (submitted as a separate report).

Fundamentally, low temperature refrigeration requires an effort in terms of work to remove heat from a low temperature and reject this heat to the surroundings at a higher temperature. The highest thermodynamic efficiency, expressed as a ratio of work to remove heat and the amount of heat removed, is that of the Carnot cycle. The Carnot cycle efficiency is expressed as follows:

$$\frac{W}{Q} = \frac{T_1 - T_2}{T_2}$$

where W is work required to remove an amount of heat Q from a temperature of T_2 and reject it to surroundings at a temperature of T_1. In the case of refrigeration at a temperature level of 75°K with surroundings at a temperature level of 300°K, the ratio is 3, i.e., one watt of refrigeration requires at least 3 watts of work.

In large refrigeration systems, it has been possible to obtain refrigeration with an efficiency of about 30 percent of the Carnot efficiency. However, in small systems it is impossible to reach or even approach this degree of efficiency because of several reasons. Mechanical systems with rotating or reciprocating parts exhibit friction which does not reduce proportionally with the size of the equipment. Heat gain through insulation becomes relatively large because the volume of the system reduces by the third power of the linear dimension, while surface area reduces by a second power of the linear dimension.

In addition to the above mentioned sources of inefficiency common to all mechanical systems, individual systems will have thermodynamic inefficiencies dependent on the particular process used. For instance, a relatively simple process for refrigeration is a Joule-Thomson process which has only a compressor, with moving parts operating at 300°K. Since the compressor is the only component with potential wear, the process is reasonably reliable. However, the thermodynamic efficiency of the cycle is very low, and consequently, a large amount of power is required to remove heat from the low temperature level.

Another inefficiency, dependent on the particular cycle used, is the irreversibility of heat exchangers. A heat exchanger will perform ideally with a zero degree temperature differential. Under this condition, minimum heat exchanger loss is experienced and the amount of refrigeration made available by either expansion engine or Joule-Thomson expansion process may be used entirely for removal of heat at the desired temperature level. If the heat exchanger has a finite temperature differential, some refrigeration is required to overcome this temperature difference, thus reducing the refrigeration available at the desired temperature level. If the refrigeration provided per unit of working fluid is small, the influence of the temperature difference of the heat exchangers becomes larger. For this reason, centrifugal expansion engines with a low expansion ratio require extremely good and relatively large heat exchangers to reduce the net effect of heat exchangers inefficiency.

2.1 Refrigeration Cycles

There are three basic cycles employed in mechanical refrigerators to provide refrigeration at low temperatures. Each of the processes is described in detail below, and the fundamental advantages and disadvantages are discussed.

2.1.1 Joule-Thomson Cycle

Figure 1 shows the basic schematic for the Joule-Thomson (J-T) cycle. This cycle is often referred to as a Simple Linde or Hampson Cycle. A compressor circulates the process fluid. Depending on the gas used, a discharge pressure up to 1000-2000 psig may be required. The high pressure gas passes through a heat exchanger and expands through an orifice. Afteı reaching steady state operation, the gas liquefies partially upon expansion. Non-condensed vapor, and vaporized liquid, returns through the heat exchanger to the compressor.

The Joule-Thomson expansion process results in cooling and liquefaction of the gas only when the temperature of the high pressure gas at the warm end of the exchanger is below the inversion point. If the temperature is above the inversion point, heating of the gas would occur upon expansion. Since the inversion temperature of hydrogen and helium is well below ambient temperature, the simple Joule-Thomson process will not provide the desired refrigeration unless precooling of the gas is accomplished.

Thermodynamically, the cycle is inefficient since no external work is performed by the gas in providing refrigeration. Reliability is coupled with compressor performance. With reliable compressor, the refrigerator will perform continually with very little maintenance. For a non-lubricated compressor, only loss of process

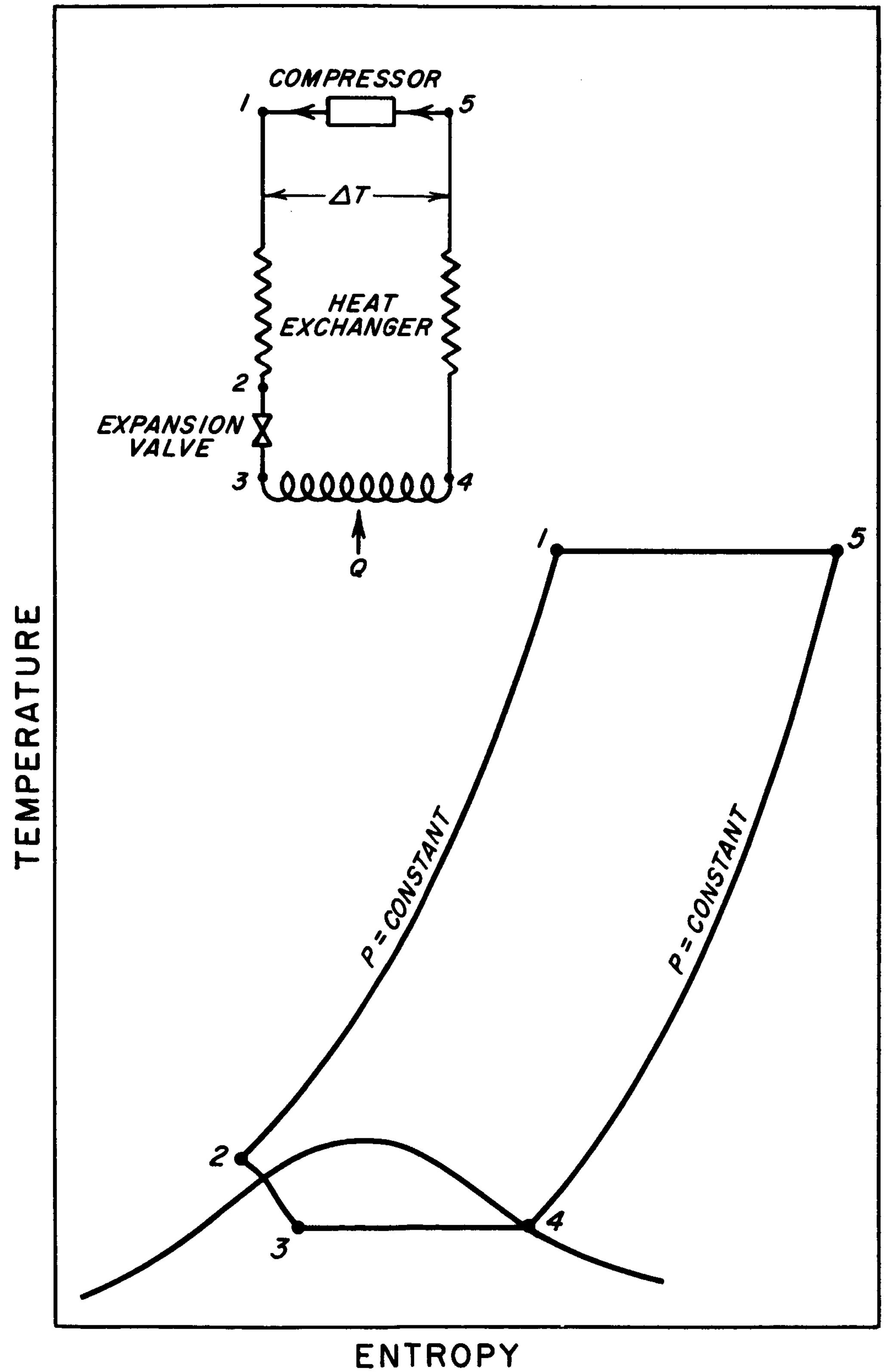

Figure 1. Joule-Thomson Refrigeration Cycle

gas from the system can cause deterioration of performance. Impurities in the process gas caused by outgassing of warm components will, in general, not cause plugging of the heat exchanger.

2.1.2 Brayton Cycle

Figure 2 shows the basic Brayton cycle. A compressor circulates gas to an expansion engine through a heat exchanger. The expanded gas provides refrigeration and returns through the heat exchanger to the intake of the compressor.

The compressor and the expander may be of the reciprocating or rotating type. In general, rotating machinery is more reliable than reciprocating machinery through the absence of rubbing parts and valves. However, a cycle consisting of a rotating compressor and expander normally is limited to a relatively low pressure ratio. The low pressure ratio provides a small amount of refrigeration per unit of process fluid circulated. Therefore, heat exchangers are relatively large and performance of the exchanger is of great significance.

Over-all thermodynamic efficiency of the process can be quite good, but is a function of the efficiency of individual components.

2.1.3 Stirling Cycle

Figure 3 shows the process for a Stirling cycle. Two cylinders are interconnected by means of regenerator and heat exchangers. The cylinder volumes increase and decrease; the change in volume of the cylinder operating at ambient temperature lags that of the low temperature cylinder by approximately 90 deg. When the

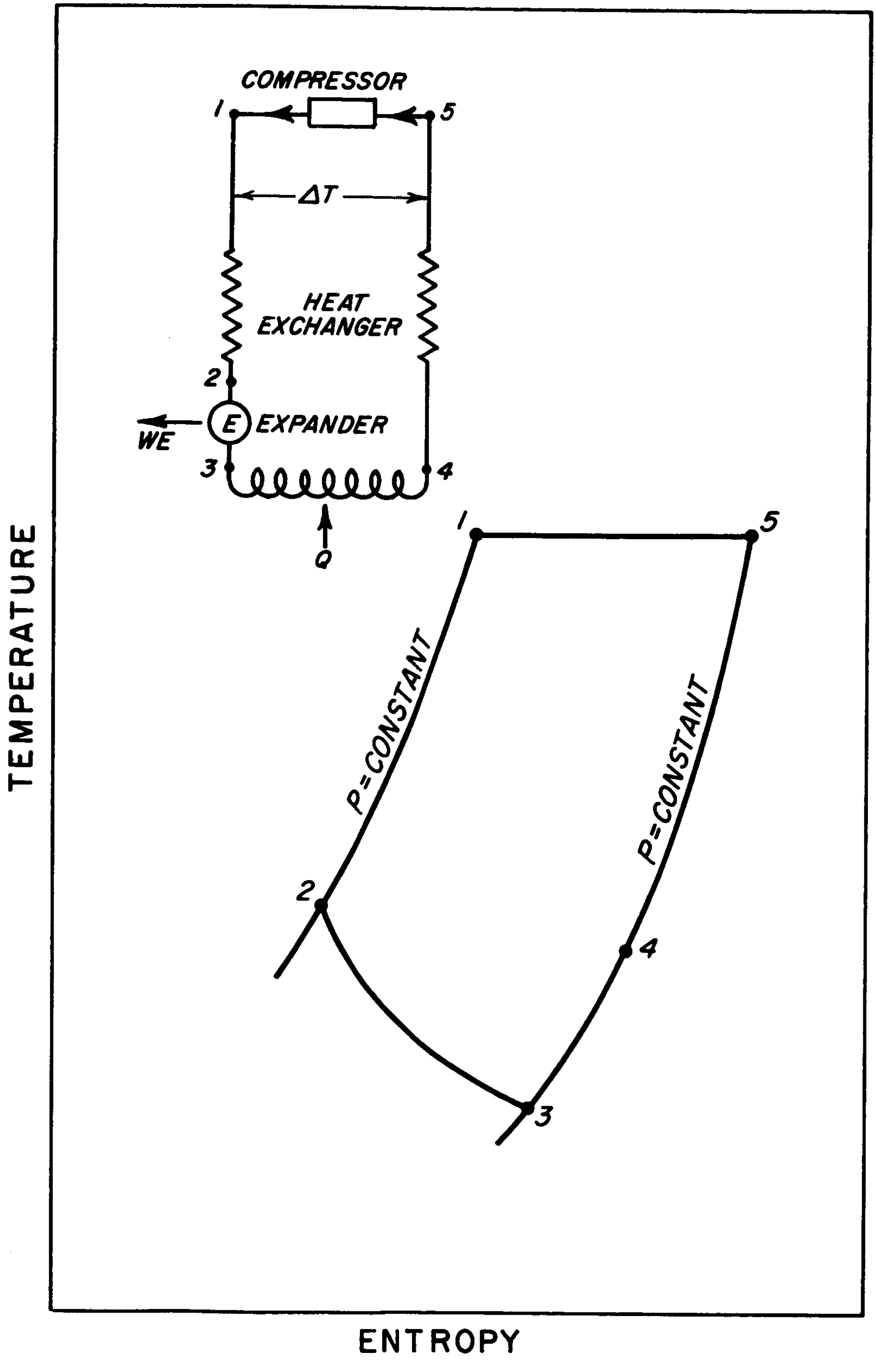

Figure 2. Brayton Refrigeration Cycle

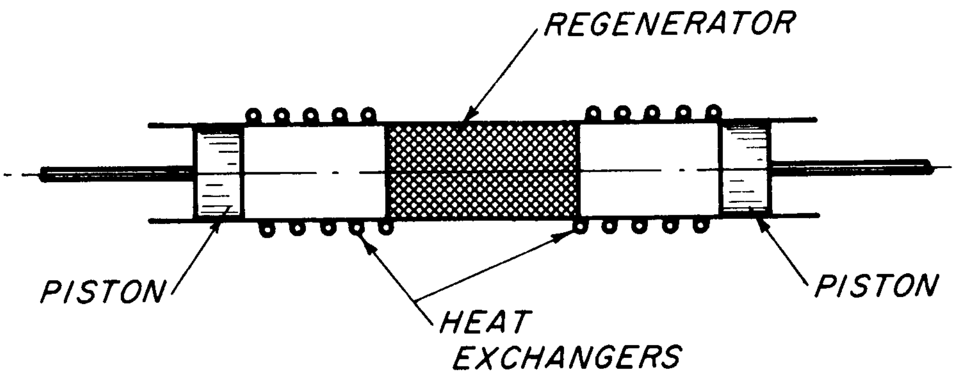

Figure 3. Stirling Refrigeration Cycle

cylinder operating at low temperature is at maximum volume, the cylinder operating at ambient temperature is still 90° away from maximum volume, but approaching maximum volume.

An analysis of the cycle indicates that, in principle, refrigeration may be obtained with the Carnot efficiency. In practice, the cycle is fundamentally as efficient as the cycle described under 2.1.2.

The physical embodiment of the Stirling cycle consists of reciprocating machinery with the absence of valves. In one particular arrangement, the so-called displacer type (Ref. 18, 26, 40), only one of the two pistons must be sealed, since the other piston has become a displacer moving gas from the cold cylinder to the warm cylinder and vice versa. Since cold and warm cylinders are in open communication with each other through heat exchangers and regenerator, very little pressure difference exists at any time between the two cylinders. As a consequence, the tolerance between cylinder wall and displacer piston is not critical.

Refrigeration is generated in the cold, or expansion cylinder, and heat from the source to be refrigerated is taken up in the heat exchanger separating the cold cylinder and the regenerator. Heat is rejected to ambient temperature in the heat exchanger separating the warm cylinder and the regenerator.

Reliability of the refrigerator is a function of the mechanical reliability of the compressor and the expander, or in the displacer-type machine, of the compressor. Therefore, the displacer-type Stirling cycle is about as reliable as a simple Joule-Thomson cycle as described under 2.1.1.

2.2 Process Components

Major components used in the three cycles described above are the following:

Compressor

Expander

Heat exchangers

Control system

Orifice or capillary

2.2.1 Compressors

Compressors may take various forms, but the main classifications are either rotary or reciprocating. In the case of reciprocating compressors, it is possible to distinguish between one or more compression stages. The Joule-Thomson(J-T) cycle described under 2.1.1 requires reasonably high pressures in order to obtain refrigeration. For this reason, compressors used for the Joule-Thomson cycle are normally mulit-stage machines with relatively high compression ratio per stage. The compression ratio per stage for the miniature refrigerator is considerably higher than that of similar compressors having very large flow rates. The use of high compression ratios is made possible through the relatively large ratio of surface area to volume for the compressor cylinder. Compression of the gas is more nearly isothermal due to heat transfer with the surface. The favorable heat transfer condition only serves a function if the heat can be dissipated from the cylinder and piston. Since wear is closely related with high temperatures of moving parts, the life of a compressor with high compression ratios is inherently less than that of a compressor with low compression ratios.

The compressor used for the cycle described under 2. 1. 2 can either be rotating or reciprocating. In the case of a rotating machine, compression ratios will be low. In order to obtain a reasonable compression ratio and high efficiency, high speeds must be employed. Bearings, therefore, become of prime importance in the design, and for this reason, current development work is concentrated in this area. When a reciprocating compressor is used for the cycle described under 2. 1. 2, a relatively low compression ratio may be used.

The compressor used for the Stirling cycle is a reciprocating machine with a low compression ratio. In fact, it is possible to reduce forces on the piston by means of maintenance of an average pressure below the piston. The compressor used for the Stirling cycle combines the features of low compression ratio and low speeds, and therefore is expected to show a good life expectancy. Experience with equipment built by various manufacturers confirms this fact.

2. 2. 2 Expanders

Expanders are either of the reciprocating or rotating type. In the case of rotating expanders, the minature size requires high rotating speeds, and as a result, high speed bearings again are the main problem area. The combination of low temperature operation of the expander proper, with ambient temperature operation of the bearings, introduces the problem of heat leak along the shaft. Operation of the bearings at low temperature has disadvantages, which in general, appear to outweigh the advantages.

Because of the combination of mechanical and thermal problems, the over-all efficiency of the miniature rotating expander will not be extremely good.

Reciprocating expansion engines do not have the same problems as the rotating expanders. It is possible to separate the mechanical support and bearing system from the cylinder and piston operating at low temperatures by means of rods of some length. However, the reciprocating expander has other problems. For instance, valves are required. These valves are timed and must seal to make efficient thermodynamic performance of the expander possible. The sealing of the piston in the cylinder also presents problems. In some expanders (for instance those used in the Collin's Cryostat) sealing is obtained through extremely close tolerances between piston and cylinder. The close tolerances introduce the problem of maintaining extreme cleanliness of the system.

In general, it can be stated that the state-of-the-art of designing, constructing and operating miniature reciprocating expanders is further advanced than that of miniature turbine expanders. Also, because of fundamental problems and difference in experience, over-all thermodynamic efficiency of the reciprocating expanders has been better than that for rotating expanders.

The Stirling cycle also uses an expander. Depending on the configuration of the mechanical parts of the refrigerator, this expander may take the form of a reciprocating machine as described above, or a displacer. The displacer performs the function of moving gas from one space to another, through a set of heat exchangers and regenerators. A more detailed description of the action of the displacer can be found in Ref. 40. The advantages of the displacer are obvious. There is very little pressure difference across the displacer and forces therefore are small. Also, because of this small pressure difference, sealing of the displacer in the cylinder is not a critical problem. An additional advantage of the Stirling cycle is the absence of valves.

On the basis of the advantages described above, and the elimination of some problems, it can be anticipated that the over-all efficiency of the Stirling cycle, even in the miniature form, will be better than that of the other cycles described. Practical experience bears out this anticipation.

2.2.3 Heat Exchangers

Very little elaboration is required concerning heat exchangers for all cycles presently under discussion. Very efficient miniature heat exchangers can be built for all cycles described. In practice, these efficient exchangers have been built. However, size and weight limitations are of major consideration for satellite requirements.

2.2.4 Control Systems

In order to operate at a constant temperature, a control system is required. If excess refrigeration is available, the temperature of the refrigerated device will be reduced. The temperature will increase when insufficient refrigeration is provided. Basically, the refrigeration provided is matched with the refrigeration load on the system. Control can be performed in a number of ways, several of which will maintain high thermodynamic efficiency.

The easiest but most inefficient method for a miniature system is the supply of additional heat from an external source. Fortunately, more efficient control systems are possible. Each particular cycle must be analyzed to determine the best method for controlling refrigeration output.

For the Joule-Thomson cycle, control of the temperature at which refrigeration is provided is exercised through selection of the process gas and the pressure at which the liquid vaporizes. The quantity of refrigeration is determined by the quantity of gas circulated through the system and the high pressure level.

Control of temperature with the Brayton cycle may be exercised through control of the quantity of gas which is circulated. Excess gas can be stored in a reservoir from where it can be returned to the cycle when required. Removal of gas reduces both work of compression and refrigeration provided by the expander.

The Stirling cycle refrigerator output may be controlled in the same manner as the Brayton cycle.

2.2.5 Orifice or capillary

Only the Joule-Thomson cycle requires an orifice or capillary to reduce the high discharge pressure of the compressor to the low pressure of the liquid reservoir. For a miniature system, it normally is advisable to combine the high pressure passage of the heat exchanger and the orifice which represents the Joule-Thomson valve. The combination then, is a capillary tube through which the pressure drop is taken over the full length of the tube.

2.3 Conclusions

The general discussion on mechanical refrigeration systems leads to the following preliminary conclusions:

a. The Stirling cycle miniature refrigerator is anticipated to have the highest over-all thermodynamic efficiency.

b. The Stirling cycle miniature refrigerator is anticipated to have the highest mechanical reliability.

c. The Brayton cycle miniature refrigerator is anticipated to have the smallest unbalanced forces and to provide operation practically free from vibration.

2.4 Review of state-of-the-art

A number of organizations were contacted and visited to discuss the state-of-the-art of miniature mechanical refrigeration equipment. A detailed discussion and evaluation of the effort made by each organization is reported separately.

3.0 Subliming Refrigeration

3.1 Basic principle

In the basic refrigerator, cooling is provided by a mass of solidified cryogen, which sublimes slowly through the supply of heat from the refrigerated object and from the surroundings. The subliming refrigerator, therefore, is a modification of a simple storage container, filled with a cryogenic liquid which evaporates. Several of the advantages of using solid as compared to liquid as a refrigerant are as follows:

a. For a certain volume, the mass of solid is larger than the mass of liquid stored, assuming that the actual density of the solid can be obtained.

b. The total quantity of refrigeration available is somewhat larger since heat of sublimation is greater than the heat of vaporization.

c. For space applications, the problems associated with zero gravity are reduced or eliminated through the use of a solid.

d. For a given cryogen, refrigeration is available at a lower temperature when in the solidified form.

3.2 Basic problem areas

a. A primary problem of the storage container used for the subliming refrigerator is obtaining adequate insulation. Data available in the literature (Ref. 43-48) indicate that the so-called super insulations or multiple-layer insulations are sufficient to make it possible to design, in principle, a relatively small container which will hold the solidified cryogen for a number of years.

A major difficulty with mulitple-layer insulation is its physical application. It is usually necessary to have pipes, supports or other penetrations of the insulation surrounding the inner container. It has been proven that discontinuities in the insulation can provide a large deterioration factor for the over-all performance of the insulation. It will be necessary therefore, to limit penetrations through and discontinuities of the insulation.

b. It will be necessary to study the method by which the inner container will be filled with the solid. A potential method to be used is to fill the container with liquid and to slowly cool and freeze this liquid while adding more liquid as necessary. Heretofore, cryogenic liquids have been solidified by use of a vacuum pump to reduce the vapor pressure below the triple point. Although this method is very simple, it does not result in obtaining a solid with a high apparent density. The resulting solid is porous since a considerable fraction of the liquid has to be vaporized to provide cooling for the solidification of the remainder of the liquid. It appears, therefore, that an external or separate cooling circuit should be used.

c. Conduction of heat from the refrigerated source to the solidified cryogen and uniform temperature distribution must be considered carefully. In the case of an infrared detection cell it is desirable to have close physical coupling between the sensor and the source of refrigeration. The physical arrangement must allow the heat to be dissipated in the solid without generating large temperature gradients.

d. Temperature control of the refrigerated object may present serious problems. Pressure control of the subliming solid may be required.

e. In addition to the major problem areas listed above, a large number of minor problems exist. These are, for instance, the suspension and shock resistance of the inner container and the maintenance of the solid while the subliming refrigerator is in the earth's atmosphere. Most of these problems are of a practical nature and can be readily solved for a specific application.

3.3 Conceptual design

To demonstrate feasibility, advantages, disadvantages, and problems associated with the subliming refrigerator, practical cases will be investigated and a conceptual design of a subliming refrigerator will be presented. The assumptions are as follows:

a. Temperature of surroundings -300°K

b. Duration of refrigeration - 3 years and 1 year

c. Refrigeration temperature - 70°K and 50°K

d. Infrared sensor area - 0.2 cm^2

Refrigeration must be supplied to the infrared sensor at a continuous and constant rate. From the above assumptions this has been estimated to be 8 milliwatts. The required refrigeration has been computed for black body thermal radiation assuming that the electrical energy dissipated in the sensor is negligible.

Figure 4 indicates the concept of the subliming refrigerator. The basic feature of the refrigerator under consideration is the use of two different solid cryogens, one of which is providing refrigeration at the desired temperature level; the other is used to shield the lower temperature cryogen from heat penetrating the insulation of the system.

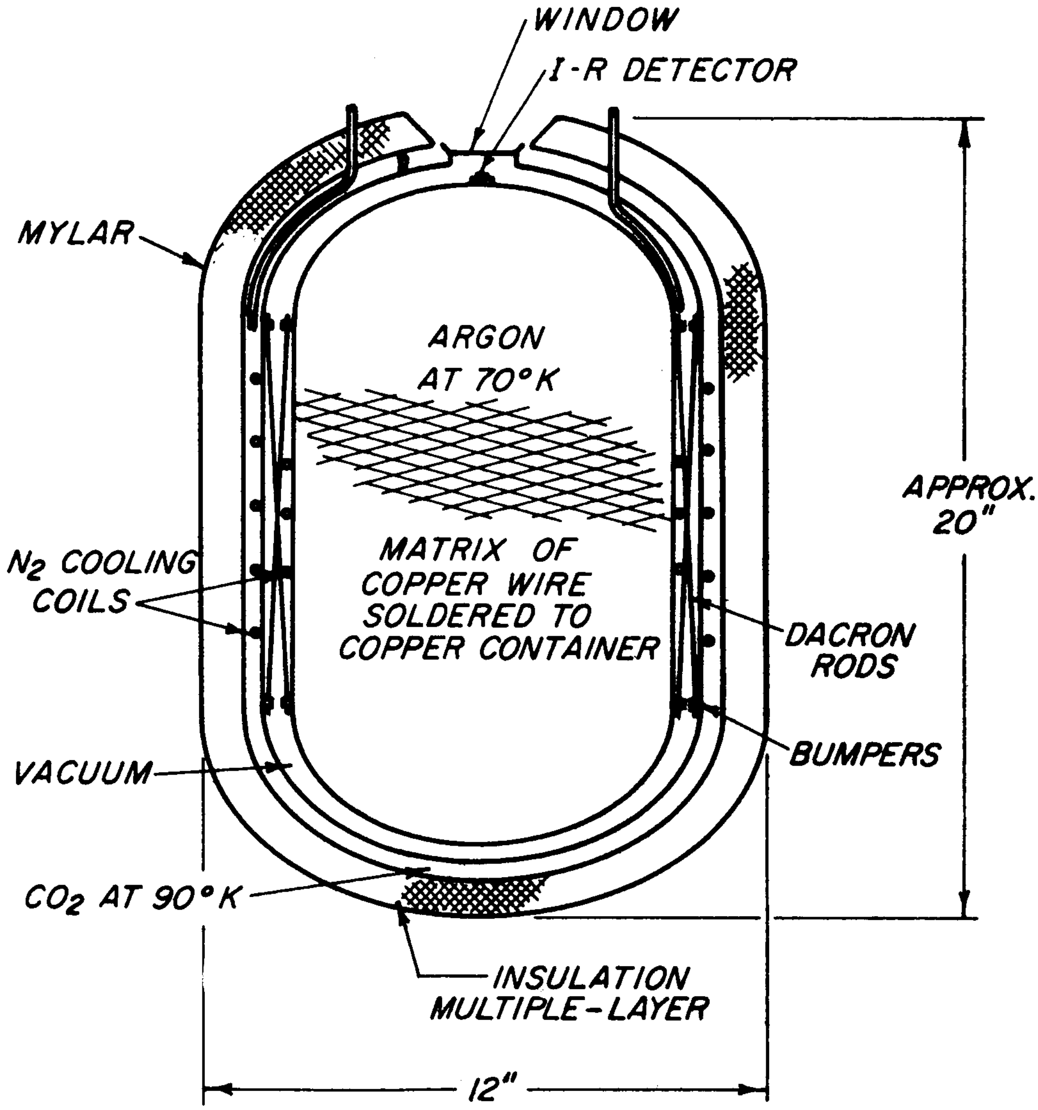

Figure 4. Argon-CO_2 Subliming Refrigerator

Selection of the shielding solid, subliming at a higher temperature, may be based on properties which will yield advantages with regard to the over-all design of the system. A study of the properties of various fluids listed in Table 1 shows that CO_2 has a high density and a high heat of sublimation on both a volume and weight basis. When the subliming refrigerator is operating at a vapor pressure of 10^{-7} mm Hg, the temperature of the solid CO_2 will be of the order of 82°K. A shield maintained at this temperature provides excellent insulation for a solid refrigerant operating at the lower temperature.

A wide choice of refrigerants is available to provide cooling of the infrared sensor. Through the use of a pressure controller, a wide range of temperatures can be maintained. For instance, when Argon is used in the solid form, maintenance of the vapor pressure of the solid at 516.84 mm Hg will provide a temperature of 83.8°K. If the vapor pressure is controlled at 10^{-7} mm Hg, a temperature of 28.5°K may be obtained.

The system Argon-CO_2 has the following advantages for a space borne subliming refrigerator.

a. The high heat of sublimation on a volume and weight basis, allows minimum dimensions.

b. Since CO_2 is in the solid form when under atmospheric pressure, the container in which the CO_2 is stored does not have to be a pressure vessel either in the earth's atmosphere or in space.

c. The temperature of solid CO_2 at atmospheric pressure is 194.7°K. A shield of this temperature provides adequate insulation for the Argon stored in the inner container of the device during the relatively short period in the earth's atmosphere.

Table 1 - Properties of Cryogens

Fluid	Molecular weight	Heat Sublimation* cal/mol	Heat Sublimation* cal/gm	Saturation Temperature* °K	Density* g/ml	Thermal Conductivity* mw/cm °K	Triple Point °K	Triple Point mm Hg
CO_2	44	6486	147.4	81.7	1.70	5.53	216.6	3885
C_2H_2	26	≈ 3400	130.7	98.6 at 10^{-3} mm	≈ 0.4		192.4	962
CH_4	16	2340.7	146.2	35.	0.522	≈ 0.01	90.64	87.68
A	39.9	1936.9	48.54	28.5	1.75	8.3	83.80	516.84
C_2H_4	28	4809.4	171.8	60.5	0.74		103.7	-
CO	28	2032.2	72.57	28.2	1.028		68.127	115.3
N_2	28	1769.0	63.18	25.1	1.02	≈ 0.045	63.152	94.12
O_2	32	2209.2	69.04	30.0	1.40	≈ 0.045	54.352	1.10

* Pressure - 10^{-7} mm Hg

d. The solid CO_2 may be insulated from ambient with multiple-layer insulation without vacuum in the earth's atmosphere. Although the rate of heat conduction through the insulation will be relatively high, the exposure time will be short. As a result, a light weight shell separating the insulation and the solid CO_2 may be used.

e. Argon and CO_2 may be solidified through the use of liquid nitrogen boiling at atmospheric pressure. Since Argon in the solid form has a high thermal conductivity, rapid freezing with the use of a simple system may be anticipated.

f. The high thermal conductivity of Argon is helpful in the dissipation of heat from the infrared sensor.

g. It is not necessary to provide multiple-layer insulation between the CO_2 cooled shield and the Argon container.

h. CO_2, Argon, and nitrogen are all inert gases and require no special safety considerations at the launch facility.

3.4 Design calculations

3.4.1 Subliming refrigerator - Figure 4 and Figure 5

Details of the conceptual design for the refrigerator configuration shown in the figure are summarized in column one of Table 2. A sensor temperature of 70°K and a refrigeration time of three years was selected as the basis for calculation.

The cooling channels (stainless steel tubes 0.1875 inch O.D.x 0.005 inch wall) provided within the vessel allow the refrigerants to be solidified at the temperature of liquid nitrogen (78°K). The final temperature of the Argon refrigerant is not obtained until after launch where further reduction in the vapor pressure can be achieved by the environment. The volume occupied by the refrigerants have, therefore, been computed at a density corresponding to 78°K.

The ultimate temperature of the CO_2 system will depend on the orbital altitude, since it is in open communication with the environment. At a pressure of 10^{-7} mm Hg, corresponding to approximately 100 mile altitude, the equilibrium temperature is 83°K. There is no requirement for a pressure control. The present calculations assume a temperature of 90°K for the CO_2.

It was assumed that the inner container surrounding the Argon refrigerant was composed of a cylindrical portion (L = D) and 2 hemispheres. Spacing of the larger shells were made concentric to the inner container. The material for the container walls was selected to be copper with a thickness of 0.010 inch except for the shell surround- the Argon insulating vacuum space. Since this shell must withstand the stress of an external load of one atmosphere, it was necessary to select a higher strength material. Calculations indicated that stainless steel (304, 316) with a wall thickness of 0.024 inch would be adequate. Both

Table 2 - Subliming Refrigerator Characteristics

	Figure 4	Figure 7	Figure 7	Figure 8
Sensor Requirements				
Temperature - °K	70	50	70	50
Duration-years	3	1	1	1
Heat load-mw	8	8	8	8
w-hrs	210	70	70	70
Refrigerator				
Primary Refrigerant	Argon	Argon	Argon	Argon
Container Geometry				
Volume-in.^{3}	293	94	80	910
Surface area-in.^{2}	242	110	100	500
Dimensions-in.	6.2D x 12.4	4.17D x 8.34	4.0D x 8.0	9D x 18
Weight- container-lb	1.0	0.6	0.5	2.0
refrigerant-lb	17.4	5.3	4.5	2.67
total-lb	18.4	5.9	5.0	4.67
Insulation	HV*	-	-	HV
Heat Transfer				
Sensor-mw	8	8	8	8
Sensor support-mw	-	-	-	5
Supports-mw	1.94	1.77	0.95	1.36
Radiation-mw	3.88	2.74	1.92	10.4
Fill and vent tubes-mw	2.06	1.25	0.69	8.25
Total-mw	15.88	13.76	11.56	33
w-hrs	418	121	101	289

Table 2 - Subliming Refrigerator Characteristics (continued)

Secondary Refrigerant	CO_2	CO_2	CO_2	Shield
Container Geometry				
Volume-annular space-in^3	268	97.3	100	
Dimensions				
O. D.-in	8D x 15.8	6.25D x 10.4		
I. D. -in	7.2D x 13.4	5.17D x 9.34		10D x 20
Weight-				
container-lb	3.56	1.45		
refrigerant-lb	16.0	5.4		
total-lb	19.56	6.85	7.0	5.4
Insulation				
Type	ML*	ML	ML	ML
Weight-lb	4.0	1.93	2.0	6.2
Heat transfer-mw	53	55		
w-hrs	1395	482		
Overall Geometry				
Dimensions-in	12.1Dx20.3	9.85D x 14.0		14.5D x 24.5
Weight-lb	42.0	14.63	14.0	16.3

* HV-high vacuum, ML-multiple layer

metal surfaces surrounding the high vacuum annulus must be highly reflective to provide a low emissivity.

Mylar was selected as the material for the outer shell surrounding the multiple-layer insulation. The adaptability of mylar for applications of this nature has been successfully demonstrated. While in the earth's atomosphere, the multiple-layer insulation can be pressurized with CO_2 gas at a pressure of one atmosphere. Thus, no differential pressure exists across the mylar. The heat transfer through the insulation is expected to be equivalent to the thermal conductivity of the CO_2 gas (at 0°C, $k = 1.28 \times 10^{-4}$ w/cm °K).

The high thermal conductivity results in a heat flux 300 times greater than multiple-layer insulation under high vacuum conditions ($k = 3.5 \times 10^{-7}$ w/cm °K) and approximately 10 times greater than multiple-layer insulation under a compressive load of one atmosphere. Nevertheless, it is adequate for the present application because of the short period of exposure time. The refrigerant can be retained before launch, without loss, by the circulation of liquid nitrogen through the cooling channels provided. After launch, the pressure in the multiple-layer insulation will decrease, without need for controls, since the annulus is open to the environment.

The weights given for the containers include the estimated weights of supports, brackets, and other required components not separately tabulated.

The support system for the inner shell is composed of six Dacron cables or rods. The choice of Dacron was based on the favorable ratio of allowable stress to thermal conductivity as compared to metals and to other non-metallic materials (Ref. 55). Three of the

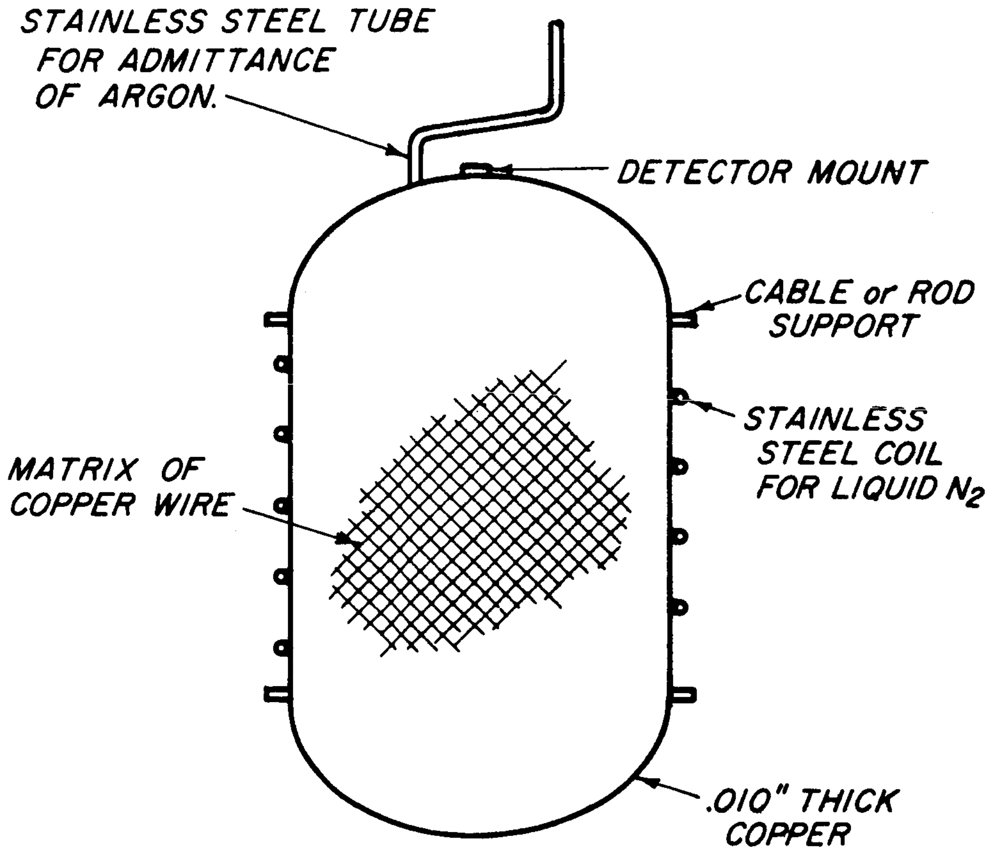

Figure 5. Argon-CO_2 Refrigerator - Inner Container

support members were computed on the basis of a 30 g loading during launch. The remaining three are required for proper positioning, and are computed on the basis of a 1 g loading.

In considering the tabulated figures, it appears that a somewhat larger amount of insulation, or an improvement in the quality of the insulation, would reduce the over-all weight of the system. A reduction in the heat transfer through the insulation by a factor of 2 will reduce the required CO_2 by 8 pounds, but will increase the required quantity of insulation by approximately 6 pounds. A more detailed parameter study of the system is required to arrive at an optimum design. The foregoing indicates what can be done in the design of a subliming refrigerator. It is suggested that a contingency factor of perhaps 10-15 percent be applied to the over-all weight of the subliming refrigerator.

A more simple system is shown in Figure 6. A single container is equipped with a coil, through which liquid nitrogen may be forced. The container is surrounded with a few inches of super insulation. The sensor is located at one end of the container and a penetration through the insulation allows radiation to pass.

To provide refrigeration for a period of three years at a temperature of approximately 85-90°K, CO_2 may be used to fill the container. The process of filling is the same as described before. CO_2 gas of one atmosphere pressure is maintained in the multiple-layer insulation. Prior to launch only moderately good insulation is provided, which if desired can be improved through the use of liquid nitrogen.

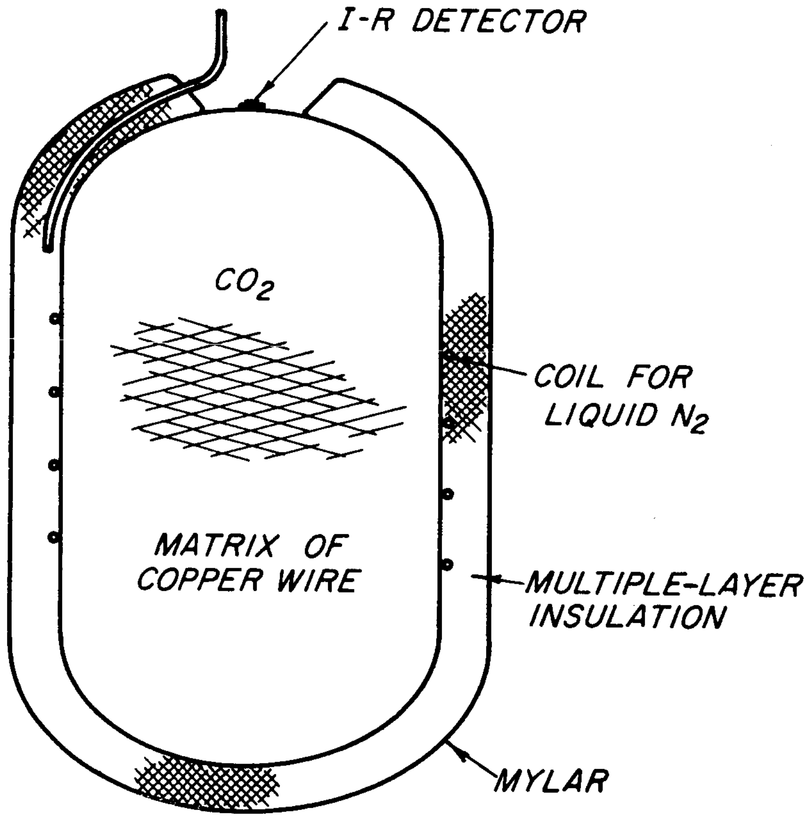

Figure 6. CO_2 Subliming Refrigerator

The advantages of the CO_2 system of Figure 6 is its extreme simplicity. A period of three years of continuous refrigeration requires less than 10 lbs of solid CO_2. The total weight of the system will be of the order of 15 lbs. Although the system is advantageous for it's simplicity and low weight, the lowest temperature obtainable, in practice, would be approximately 85°K. This presents a severe limitation for future applications.

3.4.2 Subliming Refrigerator-Figure 7

An alternate arrangement for the subliming refrigerator is shown in Figure 7. The high vacuum in the annulus surrounding the Argon container has been replaced by liquid nitrogen. The liquid nitrogen provides the cooling necessary to solidify the refrigerants and provides stand-by capabilities. At launch, the liquid nitrogen can be removed to reduce weight. Since the annulus is directly connected to the environment, its pressure will decrease as the altitude increases. In orbit, the vacuum of space is sufficient to provide effective insulation to reduce the heat transfer between the primary and secondary refrigerants to design level.

The alternate system provides additional simplicity in the design, fabrication and operation of the refrigerator. Also, the scheme offers a reduction in weight by eliminating the cooling tubes (also eliminates heat transfer associated with the cooling tubes) and by easing the requirement on the outer shell of the annulus. No differential pressure exists across the outer shell of the annulus, allowing it to be of thinner material (0.010 inch copper). One additional advantage is worthy of noting. Since the need for vacuum and pressure containers has been eliminated, it is possible for the refrigerator to fit any desired shape, a cylindrical or spherical configuration is no longer essential. It would be possible to adapt the configuration to the particular satellite requirement.

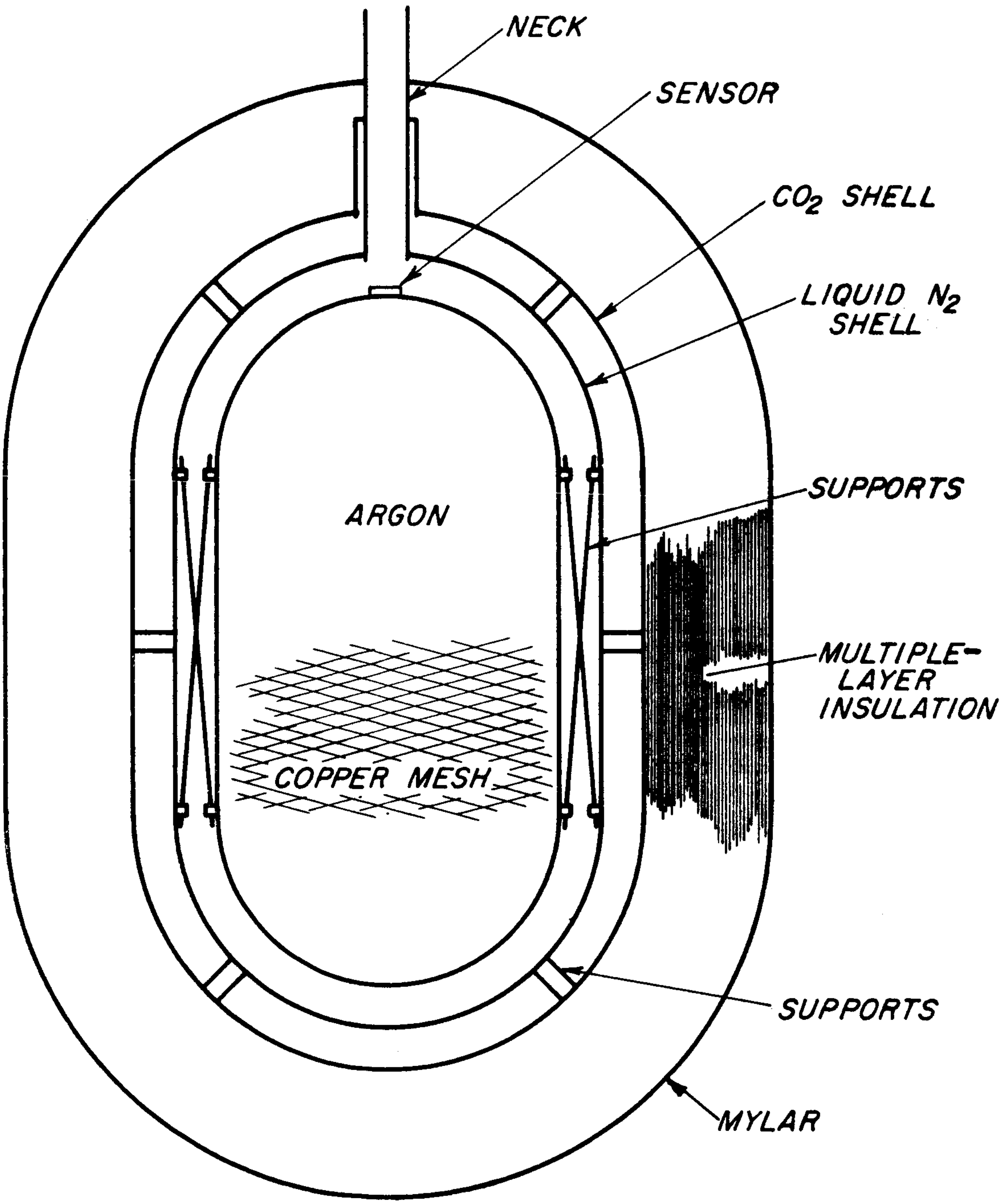

Figure 7. Argon-CO_2 Subliming Refrigerator - Alternate Design

Calculations have been performed for estimating the requirements of the alternate subliming refrigerator. Two cases were considered. The results are shown in column 2 and 3 of Table 2. Both cases have been considered for a refrigeration period of one year; one case having a refrigeration temperature of 50°K, the other 70°K. The major saving in weight is provided by the reduced quantity of primary refrigerant required.

3.4.2.1 Problem areas

Several specialized problem areas become associated with the alternate subliming refrigerator. These include the following:

a. Filling the CO_2 container may present difficulties due to the large temperature difference between the solidification temperature of CO_2 and the boiling point of liquid nitrogen. Any possibility of plugging the neck area before the CO_2 container is completely full must be avoided. It is not necessary to maintain the proposed shape of the carbon dioxide vessel. It may be located at one end of the refrigerator, with a possible reduction in weight through the elimination of part of the shell. Also, it may be possible to eliminate the shell altogether and to freeze the carbon dioxide directly in the adjoining multilayer insulation.

b. If the liquid nitrogen is not removed prior to launch, it will boil violently upon reduction in pressure and eventually freeze. The effect on the emissivities of the surrounding walls must be evaluated. Since the solid nitrogen will be at a low temperature compared to the other refrigerants, its presence may not be harmful.

3.4.3 Subliming Refrigerator-Solid Hydrogen

The conceptual design of a subliming refrigerator using solid hydrogen as a single refrigerant is shown in Figure 8. Cooling the sensor is accomplished by the flow of cold effluent gas from the hydrogen container. The sensor may be located at some distance from the refrigerator and mounted by means of a hollow stainless steel tube. The gas from the solid hydrogen container separates into two streams, one of which goes directly to the sensor. From the sensor this stream rejoins the other stream and both cool the vacuum shell and shield within the insulation. A thermometer (gas thermometer or vapor pressure thermometer) attached to the sensor will control the division of flows to maintain a constant temperature. The intermediate vapor cooled shield is essential to the efficient operation of the unit because of the large temperature difference between the solid hydrogen and the environment ($\approx$ 13.5°K to 300°K) and the low heat of sublimation of the solid hydrogen.

The two cooled shields are located within the multiple-layer insulation approximately one and two inches from the vacuum shell of the container. In the earth's atmosphere, the multiple-layer insulation can be maintained under a nitrogen gas atmosphere to eliminate the need for a vacuum shell around the insulation. Liquid nitrogen can be used to reduce the heat flux to the solid hydrogen until launch of the vehicle. Since the liquid hydrogen is intended to be admitted as liquid and solidified by the circulation of liquid helium through the cooling coils provided, it is also possible to maintain the solid hydrogen on a no-loss basis prior to launch.

The pressure maintained around the solid hydrogen is not critical. An orifice in the hydrogen tube could suffice to maintain the pressure within predetermined limits.

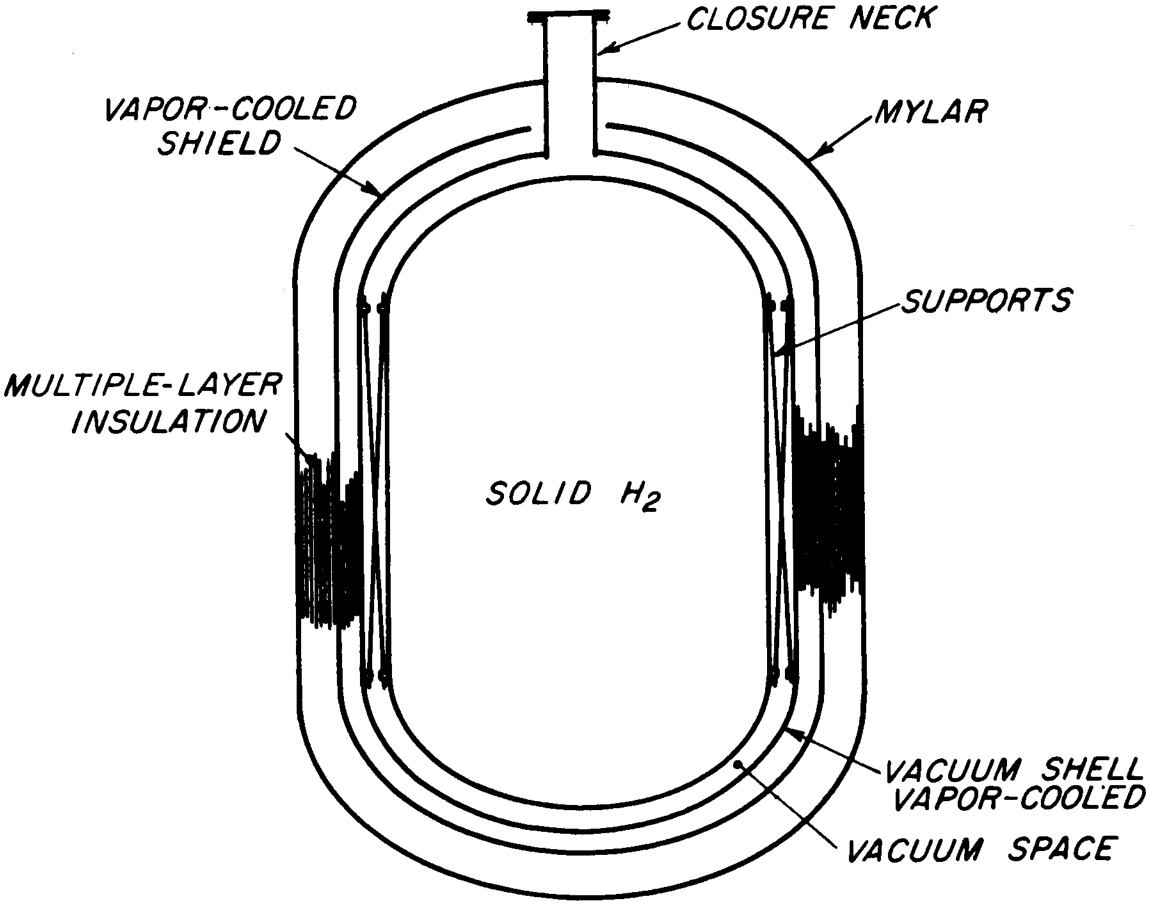

Figure 8. Hydrogen Subliming Refrigerator

To increase the amount of useful refrigeration, para to ortho conversion of the effluent gas must be accomplished at the heat sources. Suitable catalyst materials are available for such application.

The multiple-layer insulation is located around the vacuum shell. It will be noticed that there is only one penetration through the insulation, which is favorable in obtaining good over-all performance of the insulation. The neck accommodates all tubing entering the refrigerator.

The design concepts for the refrigerator utilizing solid hydrogen are the same as the previous cases considered. The results of calculations are tabulated in column 4 of Table 2. It will be noted that heat input to the sensor and connecting tube is removed by warming the hydrogen vapor and not by the heat of sublimation. The gas is evolved by the heat transfer through the container insulation.

The vacuum shell separating the solid hydrogen container and the multiple-layer insulation must withstand an external pressure of one atmosphere prior to launch. The shell has been assumed to be fabricated of stainless steel for higher strength. The solid hydrogen container is composed of 0.010 inch thick copper.

3.4.3.1 Problem areas

a. The use of liquid and solid hydrogen introduces quite considerable safety problems, especially on and around the launch facility. Solid hydrogen has a vapor pressure of less than 53 mm Hg and the system must be safeguarded from air leaking into the solid hydrogen container.

b. Filling of the refrigerator is accompanied by a number of handling problems. Liquid nitrogen, helium and hydrogen must be present on the launch facility. To be able to freeze the liquid hydrogen, a very well insulated liquid helium system must be used. The heat of vaporization of liquid helium is very low, and poor insulation of the helium system will result in excessive use.

c. Analysis of the weight distribution of the hydrogen refrigerator indicates that most of the weight of the refrigerator consists of metal and insulation. Additional investigations would result in a more optimum design with the possible saving of weight.

d. Temperature control of the sensor is more complex. A valve with variable orifice is required to maintain proper division of flow.

4.0 Comparison of CO_2 - A and H_2 Subliming Refrigerator

Table 3 compares the CO_2-A system shown in Figure 7 and the H_2 subliming refrigerator. The two systems are rated with a plus, a minus, or a zero. A plus indicates superiority over the other system, and a zero indicates approximately equal value.

Table 3

		CO_2-A	H_2
a.	Weight	0	0
b.	Volume	+	-
c.	Reliability	+	-
d.	Accessibility to detector	-	+
e.	Temperature control	+	-
f.	Temperature range	-	+
g.	Ease of manufacture	+	-
h.	Ground equipment for launch	+	-
i.	Vacuum requirements	+	-
j.	Length of development	+	-

It appears for the cases considered, that the CO_2-A system is superior to the H_2 system. It should be noted that this is only the case in the temperature range which both systems can cover. If temperature below 30-35°K are required, the CO_2-A system becomes unsuitable.

On the important variables of volume and reliability, the CO_2-A system is superior. With respect to weight, it may be noted that the CO_2-A system consists of approximately 70 percent of solid cryogen, while the H_2 system contains only 15 percent in solid cryogen.

Because of the high density of solid CO_2 and A, the CO_2-A system is always smaller in volume then a H_2 system for the same duty. Also, the CO_2-A system can be better adapted to spaces with odd dimensions. It is relatively simple to make square, rectangular or oval containers when vacuum requirements are deleted.

The CO_2-A system scores high on reliability when compared to the H_2 system. The vacuum requirement of the H_2 system, when in the earth's atmosphere, makes a more difficult fabrication process. Temperature control of the sensor is reduced to a pressure control problem in the CO_2-A system, which is not difficult down to temperatures of 50°K. In the case of the H_2 system, it is necessary to divide the effluent gas flow from the solid H_2 container which requires control valves. Temperature control is required in addition because variable heat leaks in the system outside the solid container may require variation of the division of flow.

The hydrogen system has an advantage with regard to accessibility of the sensor. The detector may be located outside of the refrigerator, and does not require insulation when in the earth's atmosphere. Accessibility of the sensor in the CO_2-A system is adequate, since penetrations through a vacuum shell are not required.

The temperature range of the hydrogen system is wider since solid hydrogen is stored at a temperature of 13.5°K or less. With proper temperature control, the sensor may be operated at a very low temperature. It should be noted that, at the lower temperatures, the arrangement of the refrigerator will change, since it will be impossible to make use of the sensible heat of the effluent hydrogen vapor for detector cooling.

The CO_2-A system is superior from the standpoint of fabrication. The refrigerator must be of vacuum quality, but can be stored and mounted in the space vehicle without the need for vacuum insulation. Vacuum leaks in the Argon container would result in loss of refrigeration capability, once the refrigerator is in space.

The ground equipment required to maintain the refrigerator in stand-by condition during launch preparation are minor for the CO_2-A system. Only a supply of liquid nitrogen is required. Safety hazards do not exist. The hydrogen system requires a supply of liquid helium with an excellent transfer system. Safety problems are considerable, since hydrogen presents potential hazards,particularly since the solid hydrogen is maintained below atmospheric pressure. Filling of the hydrogen system will require hydrogen facilities as part of the ground equipment.

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